Modeling of Oil Retention in the Suction Line and Evaporator of Air-Conditioning Systems

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Received January 17, 2005; accepted August 15, 2005

This paper presents a general method for computing the oil retained in the suction line and evaporator of an air-conditioning system. For the suction line, a new semi-empirical correlation for the interfacial shear stress friction factor is presented. The correlation agrees closely with experimental results for conventional refrigerants and CO2 and extends to and merges with previous correlations available in the literature. The correlated friction factor is within 26% relative error as compared to experimental data. In the evaporator, oil retention volume was computed by using the Premoli et al. (1971) void fraction model and applying heat transfer and pressure drop correlations for refrigerant/oil mixtures. As a result, oil retention was predicted within 21% average relative error.

INTRODUCTION

In air-conditioning systems the refrigerant carries a small amount of oil throughout the system and oil is retained in the components. The working fluid is not pure refrigerant but a refrigerant/oil mixture. The equations that model two-phase flows are more complicated or more numerous than those for single-phase flows, and usually semi-empirical correlations provide a simple method to describe the interfacial surface boundary conditions. Oil entrainment is a complex function of refrigerant and oil velocities, densities, mutual solubility, and miscibility. Geometry, orientation, and surface roughness of the pipe also affect the oil transport characteristics. The models for oil entrainment in the literature are mainly divided into two types: separated flow models for gas and liquid two-phase flows and homogeneous models for liquid and liquid single-phase two-component flows.

Abundant literature can be found for gas and liquid two-phase flows (Wallis 1969, 1970). Several researchers developed semi-empirical correlations for the interfacial friction factor using air and water and rearranging the results according to dimensionless groups. Newton et al. (1999) investigated two-phase flows of air with different liquids, such as water, kerosene, and Propar-22 (a light machine oil). Wongwises and Kongkiatwanitch (2001) suggested a new empirically correlated interfacial friction factor using air and water. They developed the correlation for gas Reynolds numbers that varied from $2 \times 10^4$ to $8 \times 10^4$ and for liquid film thicknesses that ranged from $0.10 \leq \delta / D < 0.17$. The flow pattern was annular flow with large disturbance waves that formed as the liquid flow rate increased. The droplet entrainment in the gas core regions was experimentally measured and taken into account in the model proposed by Wong-
Mehendale (1998) estimated liquid film thickness in an upward vertical suction line and verified his model with experimental data. Lee (2002) used a similar model to compute oil retention volume in the suction line of CO₂ air-conditioning systems. A new correlation was obtained for Reynolds numbers varying from $16 \times 10^4 < \text{Re}_g < 35 \times 10^4$, which was the range of gas core Reynolds numbers in CO₂ air-conditioning systems. The liquid film thickness ranged from $0.02 \leq \delta/D < 0.10$ and the suction line was horizontal. Lee also computed oil retention in a micro-channel evaporator for CO₂ air-conditioning systems, and he developed a model to estimate oil retention volume in the headers of the heat exchanger, as well.

A critical summary of the influence of oil on refrigerant heat transfer and pressure drop was given recently by Shen and Groll (2003). They described one possible approach for evaluating the properties of liquid refrigerant and oil mixtures, summarized the most common heat transfer correlations used during evaporation and condensation processes, and proposed pressure drop correlations depending on oil mass fraction in the mixture. Thome, who was one of the pioneers in this area, proposed a comprehensive thermodynamic approach to modeling the behavior of refrigerant and oil mixtures (Thome 1995). His theoretical and experimental investigations provided useful insights about the physics involved in the evaporation of a mixture.

In this paper, new semi-empirical correlations are used to model oil retention characteristics in the suction line for different refrigerant/oil mixtures. The simulations are verified with experimental data presented in Cremaschi’s (2004) PhD thesis and with previous experimental results available in the literature. Oil retention in a tube-fin evaporator using conventional refrigerants is also computed using the Premoli et al. (1971) void fraction model. The approach proposed by Thome and various semi-empirical correlations suggested by Shen and Groll were integrated in a software engineering tool used for designing heat exchangers (Jiang 2003). The design tool had been verified and discussed in a previous research paper by Jiang et al. (2002).

VERIFICATION OF THE FLOW PATTERN IN THE SUCTION LINE

Mandhane et al. (1974) mentioned that numerous studies have shown that no single theory or correlation can satisfactorily predict the pressure gradient or liquid holdup over all possible flow regimes encountered in two-phase gas-liquid flow in pipes. From the designer’s point of view, it is important to be able to predict accurately what flow pattern will occur from given input flow rates, pipe sizes, and fluid properties. Many methods have been presented in the literature for this purpose, usually in the form of two-dimensional maps in which the locations of the boundaries between flow pattern regions are based on empirical observations.

In this research, the modified Baker’s map (Baker 1954) is used to identify the flow pattern in the suction line. Several experiments were carried out by the authors, and a visualization section was set up in the suction line of an air-conditioning system. The complete, detailed description of the test stand and of the test conditions can be found in Cremaschi’s (2004) thesis. The data have been collected and plotted in the map of Figure 1 where the circles represent empirical observations of the flow pattern. The x-axis represents the corrected ratio of liquid mass flux $G_l$ to the vapor mass flux $G_v$, while the y-axis represents the corrected vapor mass flux. The map was originally developed for air-water two-phase flows in horizontal pipelines. However, the correction factors $\lambda$ and $\psi$ allow researchers to compensate for different properties and to apply the map for other fluid mixtures. The correction factors are given as follows:

$$\lambda = \left( \frac{\rho_l}{\rho_g} \right)^{\frac{1}{3}} \left( \frac{\rho_l}{\rho_w} \right) ; \quad \psi = \left( \frac{\sigma}{\mu} \right) \left( \frac{\rho_l}{\rho_w} \right)^{\frac{1}{3}}$$

(1)
The subscripts $a$ and $W$ refer to the values of properties for air and water at atmospheric pressure. In the map of Figure 1, the x- and y-axes were corrected according to the relations

$$X = \frac{G_l}{G_g} \cdot \lambda \cdot \psi \quad \text{and} \quad Y = \frac{G_g}{\lambda}.$$  \hspace{1cm} (2)\text{\hspace{1cm}}

As shown in the map, the mixture flow pattern in the suction line was annular flow for all data points used to verify the model. Visual inspections of the flow confirmed the flow patterns suggested by the map. Examples of annular flows for R-410A/POE mixtures in a horizontal suction line are shown in Figure 2 at oil mass fractions (OMF) that varied from 0.5 to 5.2 wt.%. By using a special dye that dissolves into the oil and adopting ultraviolet light, the presence of the oil in the flow was enhanced. More details of the liquid film motion were observed by recording dynamic videos during the experiments, and the liquid film could be seen flowing along the internal surface wall of the pipe. Thus, in the suction line, it is assumed that the flow pattern is annular flow.

**OIL RETENTION MODELS**

**Oil Retention Models of the Suction Line**

Simple analytical models that take no account of the details of the flow can be quite successful, both for organizing experimental results and for predicting design parameters. In a separated-flow model, the phases are assumed to flow side by side. Separate equations are written for each phase, and the interaction between the phases is considered. The price that is paid for greater accuracy in prediction of results is an increase in complexity. The Navier-Stokes equations can be applied to the two-phase flow in the suction line, and it is possible to look for an analytical solution. Coutris et al. (1989) successfully proved this approach and showed that analytical and numerical solutions exist. In this paper, circumferential secondary flows as described by James et al. (1987) are neglected. Phase inversion instabilities as described by Brauner and
Ullman (2002) are also neglected. Thus, finding the liquid film thickness of separation between the two phases appears trivial but ultimately is not because the governing equations are coupled to each other and are nonlinear. The adiabatic annular two-phase flow, which usually occurs in the suction line, will be described next. It is a first-order engineering approach that leads to empirical correlations useful in air-conditioning applications.

First, the Navier-Stokes equations are applied to both fluid phases. The following assumptions simplify the problem further:

- Axial symmetric flow.
- Steady-state, adiabatic, and fully developed flow.
- Oil droplet entrainment is neglected.
- The properties of the oil and liquid refrigerant mixtures are uniform along the liquid film thickness. The oil film has some refrigerant dissolved in it. Since the liquid properties are not the same as those of the pure oil, properties of oil with refrigerant solution are estimated based on the solubility data from the manufacturers.

Consider the two-phase separated annular laminar pipe flow shown in Figure 3. The oil retention volume in the suction line can be calculated by integrating the liquid film thickness $\delta$ with respect to the cross-sectional area $A$ as well as the entire length $L$ of the suction line tube. If $\delta$ is constant through the entire length of the tube, then the integral results as follows:

$$OR = \frac{\rho_{\text{liq}}}{\rho_{\text{oil}}} \cdot \left(1 - s_{\text{ref}}\right) \cdot L \cdot A \cdot \left[1 - \left(\frac{\delta}{R}\right)^2\right] \quad (3)$$

where $s_{\text{ref}}$ is the solubility of the refrigerant and oil pair, that is, the ratio of the mass of refrigerant dissolved to the mass of pure oil at a given temperature and pressure. The properties of the mixture were estimated by averaging pressure and temperature in the test section. The problem of estimating oil retention in the suction line is reduced to the problem of finding the liquid film thickness $\delta$ in the pipe at given refrigerant pressures, temperatures, and mass flow rates.

Figure 2. Flow visualization for R-410A/POE mixture in horizontal suction line.
Following Feng and Klausner’s (1997) approach, a force balance on a cylindrical element of the liquid film in which acceleration is ignored leads to the following equation:

$$\tau = \tau_i \left( \frac{R - \delta}{r} \right) + \frac{1}{2} \frac{dp}{dz} + \rho g z \cdot \left( \frac{(R - \delta)^2 - r^2}{r} \right)$$

(4)

where $\tau$ is the viscous shear stress, $dp/dz$ is the pressure gradient along the axial direction, and $\tau_i$ is the interfacial shear stress. The shear stress $\tau$ represents the viscous resistance that the oil film exerts on the refrigerant, while $\tau_i$ expresses the effect of the driving force for oil transport due to the refrigerant mass flux and viscosity. The above equation also takes into account the pressure gradient due to the flow and the gravitational effects that act on the liquid film.

For steady-state, incompressible flows, the continuity equation and the $z$-momentum equation for the liquid film region are

$$\frac{\partial}{\partial r}(ru_r) = 0 \quad \text{and} \quad \frac{\mu_f}{r} \frac{\partial}{\partial r} \left( \frac{\partial u}{\partial r} \right) = \frac{dp}{dz} + \rho g z .$$

(5)

The no-slip condition at the wall is applied and the above equations for Newtonian fluids are integrated. Finally, the following system of equations describes the liquid film and the core regions:

**Liquid Film**

$$\mu_f \frac{d u_l}{d r} = \left( \frac{dp}{dz} + \rho g z \right) \frac{r}{2} + \frac{C_1}{r} \quad R - \delta < r < R$$

(6)

**Core Region**

$$\mu_c \frac{d u_c}{d r} = \left( \frac{dp}{dz} + \rho g z \right) \frac{r}{2} + \frac{C_2}{r} \quad 0 < r < R - \delta$$

(7)
The following boundary conditions need to be satisfied:

1. Symmetry boundary condition: \( \frac{du_c}{dr} \bigg|_{r = 0} = 0 \)

2. Core-film interfacial shear stress condition: \( \left[ \frac{du_c}{dr} \right]_{r = R - \delta} = \left[ \frac{du_i}{dr} \right]_{r = R - \delta} \)

3. No-slip condition at the wall: \( u_i \bigg|_{r = R} = 0 \)

4. No-slip condition at the core-film interface: \( u_i \bigg|_{r = R - \delta} = u_c \bigg|_{r = R - \delta} \)

The oil mass flow rate can be obtained by integrating the velocity profiles over the cross-sectional area and applying the boundary conditions:

\[
m_o = (1 - \varepsilon_{ref}) \cdot \left\{ \frac{2\pi \rho_c u_c (R - \delta) + \frac{(R - \delta)^2}{2} \left( \frac{dp}{dz} + \rho g z \right) + \frac{2\pi \rho_i (R^2 - (R - \delta)^2)}{4 \mu_l} - \frac{(R - \delta) \ln \frac{R}{R - \delta}}{2} \right\} \nonumber
\]

\[
\cdots \frac{\pi \rho_i (\frac{dp}{dz} + \rho g z) (R^2 - (R - \delta)^2)}{8 \mu_i} \nonumber
\]

(8)

When the force balance applied to the gas core is drawn, the following equation for the pressure gradient is found:

\[
\frac{dp}{dz} = -\frac{4 \tau_i}{D - 2 \delta} - \rho g z \quad (9)
\]

where the pressure gradient of the core is a function of liquid film thickness \( \delta \) and interfacial shear stress \( \tau_i \). The void fraction \( \alpha \) is defined as follows:

\[
\alpha = \frac{A_c}{A} = \left( \frac{D - 2 \delta}{D} \right)^2 \quad (10)
\]

The interfacial shear stress is approximated as follows:

\[
\tau_i = \frac{1}{2} \rho_c \rho u_c^2 \quad (11)
\]

The interfacial friction factor \( f_i \) in Equation 11 is the most important parameter in estimating the liquid film thickness. Equation 11 is the classical relationship between the interfacial shear stress and interfacial friction factor proposed in the literature, and it represents a general definition for the interfacial shear stress. The friction factor is not constant and it depends on the film thickness, flow regime (laminar, transitional, or turbulent flow), and thermodynamic properties of the gas and the liquid. For given fluid properties, tube diameter, and oil mass flow rate, the system of Equations 8, 9, 10, and 11 contains five unknown quantities, which are liquid film thickness, pressure gradient \( dp/dz \), void fraction \( \alpha \), interfacial shear stress \( \tau_i \), and interfacial friction factor \( f_i \). The latter is usually obtained from empirical correlations produced by suitable experimental tests. This approach is an empirical, functional, and useful method for solving a complex two-phase flow problem using a simplified analytical model.

The above-mentioned system of nonlinear equations forms an ill-posed system with the system determinant being nearly singular. From the experimental data, the geometry of the suction line, the pressure and temperature of the mixture, and the oil retention were measured. Thus, the
properties of the refrigerant and oil mixture were calculated, and an average liquid film thickness \( \delta \) was found by using Equation 3. The average liquid film thickness \( \delta \) was inserted into the previous equations and the system of equations was solved by using the Engineering Equation Solver (EES) software (F-Chart 2004). As a result, the friction factor \( f_i \) was obtained from the solution of the system of equations. The uncertainty on the measured oil film thickness \( \delta \) was approximately \( \pm 12\% \). From the measurement accuracies and the average uncertainty on \( \delta \), the maximum uncertainty of the experimental friction factor was computed in EES. The experimental friction factor has an overall uncertainty of about \( \pm 26\% \). Additional details regarding the experimental determination of the oil film thickness and friction factor, as well as the past experimental work on oil retention, are available in Cremaschi (2004).

The approach used in the current paper was the following:

- All the liquid and vapor mixture properties were estimated using the manufacturer’s data or correlations. Values of the liquid film viscosity, density, and surface tension for different refrigerant and oil mixtures are summarized briefly in Appendix A for reference and comparison.
- Liquid film thickness and oil and refrigerant mass flow rates were measured in the experimental part of this research (Cremaschi 2004).
- The system of Equations 8, 9, 10, and 11 was solved using EES and the experimental friction factors were obtained.
- The experimental friction factors were correlated to different parameters, such as liquid film thickness, refrigerant vapor Reynolds number, and mixture Weber number. The dimensionless refrigerant vapor Reynolds number and the mixture Weber number are defined as
  \[
  \text{Re}_g = \frac{G_g D}{\mu_g} \quad \text{and} \quad \text{We}_m = \frac{G^2 D}{(\sigma_l \rho_l)},
  \]
  respectively, where \( G \) is the total mixture mass flux, \( G_g + G_l \).
- New correlations were developed for horizontal and upward vertical suction pipes.
- Using the new correlations, oil retention was predicted and compared with experimental data.

**Oil Retention Models of the Evaporator**

The thermodynamic approach is used to describe the behavior of refrigerant and oil mixtures in the evaporator. This method considers the effects of oil on the boiling temperature, specific heat, enthalpies, and pressure drop. It will be described next in the same order in which it was implemented in the heat exchanger design software (Jiang et al. 2002).

**Heat Transfer Correlation for Refrigerant/Oil Mixture Flow Boiling.** Each tube of the heat exchanger was divided into a certain number of segments and local heat transfer coefficients were determined for each segment individually. In each segment, a mean overall heat transfer coefficient was determined for the evaporating fluid and the heat source fluid. In the software tool developed by Jiang (2003), the overall heat transfer coefficient in each segment is given by the equation

\[
\frac{1}{UA} = \frac{1}{h_{\text{ref}} A_{t,in}} + \frac{(D_{out}-D_{in})}{k(A_{t,in}+A_{t,out})} + \frac{R_c}{A_{t,out}} + \frac{R_f}{h_{\text{air}} A_{t,total} \eta_0}.
\]  

By setting the heat exchanger dimensions, materials, and air-side conditions, the software tool computed the local air-side heat transfer coefficient, fin efficiency, contact and fouling resistances, and conduction resistance (Jiang 2003). One of the advantages of using Jiang’s software tool was represented by built-in correlations for tube-fin heat exchangers, which allowed a more accurate solution to be achieved. The only term that was modified in Equation 12 was the local
heat transfer coefficient of the refrigerant side. Since two-phase heat transfer and pressure drop correlations are functions of vapor quality in each segment, the local heat transfer coefficient for the refrigerant/oil mixture was computed according to the equation

\[
\frac{h_{TP,mix}}{h_L} = \frac{1}{X_{tl}},
\]

(13)

where \(X_{tl}\) is the Lockhart-Martinelli parameter \((\text{Lockhart and Martinelli 1949})\) of the refrigerant and oil mixture and \(h_L\) is the liquid-phase heat transfer coefficient based on Equation 14 using the liquid mixture viscosity:

\[
h_L = 0.023 \cdot \left(\frac{k_{ref,liq}}{D_h}\right) \cdot \left(\frac{G_{tot} \cdot (1 - x_{mix}) \cdot D_h}{\mu_{mix,liq}}\right)^{0.8} \cdot \left(\frac{c_p_{ref,liq} \cdot \mu_{mix,liq}}{k_{ref,liq}}\right)^{0.4}
\]

(14)

The semi-empirical constants \(c\) and \(n\) are specific to the oil mass fraction as indicated in Table 1.

**Void Fraction Model.** In order to compute the oil retention in the heat exchanger, the most important quantity that needs to be estimated is the refrigerant vapor over the total mixture void fraction \(\alpha\) in each segment of the heat exchanger tube.

Empirical models for the void fraction can be found in the literature. In the current work, four methods have been used and compared with the experimental data: Martinelli’s void fraction model for smooth tubes, Lockhart-Martinelli’s \((1949)\) correlated model, the void fraction model of Premoli et al. \((1971)\), and Chisholm’s \((1972)\) void fraction model. Other void fraction models can be considered, such as Turner and Wallis’ void fraction model, which was used by Hewitt and Hall-Taylor \((1970)\). These correlations were recommended by several authors in the most recent literature as the best methods for computing the void fraction during evaporation and condensation processes \((\text{Lee 2002; Shen and Groll 2003})\). Lee \((2002)\) found that the Premoli void fraction model was the most accurate method for predicting the oil retention in carbon dioxide air-conditioning systems. In the current work, the model that best fit the experimental data was Premoli’s void fraction model. It predicted the oil retention for all refrigerant/oil mixtures quite successfully, and it is described in more detail next.

Premoli et al. \((1971)\) developed an empirical correlation to estimate the void fraction in separated two-phase gas and liquid flows. When considering refrigerant and oil, the solution properties are used in Premoli’s model to describe the liquid phase while pure refrigerant vapor properties are used for the gas phase. The revised Premoli void fraction model calculates the void fraction \(\alpha\) from the slip ratio \(S\) of the refrigerant/oil mixture, which is defined by the equation

\[
S = \frac{v_{ref,vap}}{v_{mix,liq}} = \frac{\rho_{mix,liq} \cdot x_{mix} \cdot (1 - \alpha)}{\rho_{ref,gas} \cdot (1 - x_{mix}) \cdot \alpha},
\]

(15)

**Table 1. Coefficients \(c\) and \(n\) as Functions of the Oil Mass Fraction in the Refrigerant/Oil Mixture Heat Transfer Correlation Presented by Chaddock and Murther \((\text{Shen 2003})\)**

<table>
<thead>
<tr>
<th>Oil Mass Fraction [wt.%]</th>
<th>Constant (c)</th>
<th>Constant (n)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.0</td>
<td>3.90</td>
<td>0.62</td>
</tr>
<tr>
<td>1.0</td>
<td>4.72</td>
<td>0.59</td>
</tr>
<tr>
<td>2.9</td>
<td>4.36</td>
<td>0.60</td>
</tr>
<tr>
<td>5.7</td>
<td>4.97</td>
<td>0.59</td>
</tr>
</tbody>
</table>
where \( v_{ref, vap} \) and \( v_{mix, liq} \) are the superficial velocities of the refrigerant vapor and of the oil and liquid refrigerant mixture, respectively.

The liquid-phase concentration of oil circulating in a refrigeration system is a function of its location in the system. Thus, to unequivocally define this concentration, a point is chosen where all the circulating fluid is in the liquid phase, which occurs in the refrigerant line between the condenser and the expansion device. The oil mass fraction at this location is defined as the absolute oil mass flow rate fraction \( \omega_0 \):

\[
\omega_0 = \frac{m_{oil}}{m_{oil} + m_{ref}} = \frac{OMF[\text{wt.\%}]}{100} \tag{16}
\]

After the expansion device, the local vapor quality will range from 0.10 to 0.30 and the oil mass fraction in the liquid phase will have increased. Along the evaporator, the oil mass fraction continues to rise as the refrigerant evaporates into the vapor phase. The local mixture quality \( x_{mix} \) is defined by the mass flow rate of refrigerant vapor divided by the total mass flow rate of refrigerant and lubricant:

\[
x_{mix} = \frac{m_{ref,g}}{m_{tot}} = \frac{m_{ref,g}}{m_{ref,g} + m_{ref,l} + m_o} \tag{17}
\]

The vapor pressure of oil is negligible compared to that of refrigerants. Consequently, the lubricants can be assumed to have no effects on the mixture vapor pressure (Shen and Groll 2003). From the conservation of mass of the two components, and assuming no oil enters the vapor phase, the following expression relates local oil mass fraction \( \omega_{local} \) to local vapor quality \( x_{mix} \) and inlet oil mass fraction \( \omega_0 \):

\[
\omega_{local} = \frac{\omega_0}{1 - x_{mix}} \tag{18}
\]

The above equation is based on steady-state flow of the mixture through the system, i.e., there is no local oil holdup in the heat-transfer tubes. Since the oil is assumed to be nonvolatile, the maximum exit vapor quality that can be achieved in the evaporator is

\[
x_{mix,max} = \frac{m_{ref, gas}}{m_{ref, gas} + m_{oil}} = 1 - \omega_0 \leq 1 .
\]

If the refrigerant and oil mixture reaches \( x_{mix,max} \), then the heating fluid can increase the mixture temperature without increasing the mixture quality, as occurs in single-phase heat transfer processes. Even though the mixture quality is less than 1, this situation is referred to as superheating of refrigerant because all the liquid refrigerant has been evaporated and superheated. Moreover, according to the exit pressures and temperatures, a minimum amount of refrigerant remains dissolved in the oil and does not participate in the evaporation process.

Premoli’s correlation, usually known as the CISE correlation, covers a reasonably wide range of data and takes account of the mass flux effects (Hetsroni 1982). When immiscibility occurs, two liquid phases coexist: one is an oil-rich phase, and the other is a refrigerant-rich phase. In this case, cloudiness and/or particles may disperse throughout the liquid mixture. In totally immiscible refrigerant/oil mixtures, the lubricant and the refrigerant can be treated separately. Shen and Groll (2003) stated that the immiscibility is likely to occur in the evaporator, which
leads to the fact that more oil is trapped in the evaporator than in the condenser. In a partially miscible mixture, when the temperature is higher than the critical solution temperature, the refrigerant is totally miscible with the lubricant. When the temperature is lower than the critical solution temperature, the miscibility is dependent on concentration at a given pressure. With respect to some new refrigerant/oil pairs, such as R-410A/POE, immiscibility may occur at both high and low temperatures. Premoli’s model should not be employed for immiscible refrigerant/oil pairs because it accounts for average liquid film properties. As a first approximation, Premoli’s void fraction model was used for R-410A/MO mixtures by substituting for the average liquid film viscosity and surface tension with an estimated increased viscosity and surface tension (as is described next). Furthermore, R-134a/PAG and R-410A/POE are partially miscible mixtures and the liquid film viscosity and surface tension were also increased to account for local behavior according to the following equations:

\[
\mu_{\text{local}} = \gamma_{\text{misc}} \cdot \mu_{\text{oil}} + (1 - \gamma_{\text{misc}}) \cdot \mu_{\text{liq,average}}
\]

\[
\sigma_{\text{local}} = \gamma_{\text{misc}} \cdot \sigma_{\text{oil}} + (1 - \gamma_{\text{misc}}) \cdot \sigma_{\text{liq,average}}
\]

where \(\gamma_{\text{misc}}\) is an empirical coefficient ranging from \(0 < \gamma_{\text{misc}} \leq 1\), and it depends on the degree of miscibility between refrigerant and oil pairs. For the evaporator, \(\gamma_{\text{misc}}\) is given in Table 2 for all refrigerant/oil mixtures tested during the experimental part of this research.

The other liquid mixture properties were computed according to the general thermodynamic approach summarized by Shen and Groll (2003). Figure 4 schematically shows the oil retention model at each segment of the evaporator. From the inlet conditions, the local oil mass fraction \(\omega_{\text{local}}\) is computed first according to the above-mentioned relations. Thus, the local properties of the mixture, the Lockhart-Martinelli parameter \(X_{\text{tl}}\), the local two-phase heat transfer coefficient of the mixture \(h_{\text{TP,mix}}\), and the void fraction of the mixture \(\alpha\) in each segment were calculated accordingly. The software estimated the local heat transfer coefficient of the air side \(h_{\text{air,side}}\) so the overall heat transfer coefficient \(UA\) of the segment could be calculated (CEEE 2004). The \(\varepsilon\)-NTU method predicted the enthalpy difference across the segment, while the refrigerant pressure drop was computed using refrigerant/oil correlations. The relation among enthalpy \(h_{\text{mix,quality}}\), and bubble temperature \(T_{\text{bub,out}}\) of the mixture was determined by the heat release enthalpy curve defined by Thome (1995) and included in Appendix B. Oil retention in each segment, \(\text{ORM}_{\text{segment}}\), was estimated using the Premoli void fraction model, and these values were then summed to calculate the oil retention in the entire evaporator:

\[
\text{ORM}_{\text{evaporator}}[\text{g}] = \sum_{\text{tube}} \left( \sum_{\text{segment}} (\omega_{\text{local,segment}} \cdot V_{\text{segment}} (1 - \alpha_{\text{segment}}) \cdot \rho_{\text{mix,liq,segment}}) \right)
\]

### Table 2. Empirical Constant \(\gamma_{\text{misc}}\) to Account for Immiscibility Effects in the Premoli et al. (1971) Void Fraction Model During Refrigerant/Oil Mixture Evaporation

<table>
<thead>
<tr>
<th>Refrigerant/Oil Mixture</th>
<th>Degree of Miscibility</th>
<th>(\gamma_{\text{misc}})</th>
</tr>
</thead>
<tbody>
<tr>
<td>R-22/M0, R-134a/POE</td>
<td>Completely miscible</td>
<td>1.0</td>
</tr>
<tr>
<td>R-410A/POE</td>
<td>Partially miscible</td>
<td>0.9</td>
</tr>
<tr>
<td>R-134a/PAG</td>
<td>Partially miscible</td>
<td>0.85</td>
</tr>
<tr>
<td>R-410A/MO</td>
<td>Almost immiscible</td>
<td>0.1</td>
</tr>
</tbody>
</table>
The average deviation and standard deviation were computed according to the following equations:

$$\sigma_{avg} = \frac{1}{N} \sum \left( \frac{OR_{measured} - OR_{calculated}}{OR_{calculated}} \right) \times 100$$

and

$$\sigma_{standard} = \left[ \frac{n \sum \sigma_{avg}^2 - (\sum \sigma_{avg})^2}{n \cdot (n - 1)} \right]^{0.5}$$

The average deviation provides the error in the oil retention predicted by the heat exchanger design software, while the standard deviation is a measure of how widely values are dispersed from the mean value.

**SIMULATIONS AND DISCUSSION**

**Oil Retention in the Suction Line**

The friction factor is needed to close the system of equations presented above and to solve for the oil retention. Various friction factor correlations were used in an attempt to accord closely with the data. Figure 5 shows the variation of the interfacial friction factor with the dimensionless liquid film thickness in the suction line. The Lee (2002) and Wongwises and Kongkiatwanitch (2001) correlations predicted the friction factor characteristics better. The latter, however, was developed for lower Reynolds numbers and higher liquid film thicknesses compared to those observed in the experimental part of this research. Lee developed an empirical correlation for CO₂ and PAG oil mixtures but did not consider the solubility effects. In his work, the properties of oil and oil retention characteristics were estimated without accounting for the mass of CO₂ dissolved into the oil. After taking this effect into account in Lee’s data, the experimental friction factors for CO₂/PAG oil mixtures were plotted (Figure 5). The results show that the CO₂/PAG oil mixture has a higher liquid film thickness and lower interfacial friction factor than conventional refrigerants. The main deviation was observed at very low liquid film thicknesses.

Figure 4. Oil retention model in the heat exchanger tubes.
New semi-empirical correlations for all refrigerant and oil mixtures are proposed based on experimental results in the suction line. The newly developed correlations represent an extension of the Wongwises and Kongkiatwanitch correlation for low liquid film thickness.

Figure 6 shows the variation of the friction factor $f_i$ with the refrigerant vapor Reynolds number for an R-22/MO mixture in a horizontal suction line; $f_i$ is plotted versus $Re_g$ at different dimensionless liquid film thicknesses, which varied from 0.0032 to 0.013. As can be seen, for constant values of $\delta/D$, as $Re_g$ increased from 22,000 to 40,000, the interfacial friction factor decreased. Figures 5 and 6 indicate that changes in the refrigerant vapor Reynolds number and dimensionless liquid film thickness have an effect on the interfacial friction factor. With this in mind, the significant variables for an empirical correlation obtained from the present study are considered to be the dimensionless liquid film thickness $\delta/D$, the refrigerant vapor Reynolds number $Re_g$, and the mixture Weber number $We_m$. More complex forms of the friction factor were developed, but they increased the accuracy of the results less than 1%. Furthermore, as a result of the increased complexity of the correlation, the system of equations to predict oil retention became more and more singular. The model was developed initially for R-22/MO mixtures because it is the most common refrigerant/oil mixture used in the air-conditioning industry. A more general correlation was developed for prediction of oil retention with other refrigerant and oil mixtures to verify that the model behaves as the experiments do. The correlations that fit the data best and that were simple but accurate enough to be employed for practical use were the following.

1. **R-22/MO mixture in horizontal suction line:**

$$f_i = 8.915 \times 10^{-6} \cdot Re_g^{0.365} \cdot \left(\frac{\delta}{D}\right)^{-1.273} \cdot We_m^{0.512}$$

(23)
Equations 23 and 24 were developed and verified only for R-22/MO mixtures in suction lines with pipe diameters ranging from 12.7 to 19 mm. Equation 25 is valid only for horizontal suction lines, and it includes R-22, R-410A, and R-134a mixtures. Equation 25 is plotted in Figure 5. It fits closest the experimental data for conventional refrigerants obtained from the experimental part of this research and estimates the friction factor of CO₂ and PAG-oil mixture obtained from Lee’s experimental results quite well. As can be seen, the interfacial friction factor is inversely proportional to the dimensionless liquid film thickness, $Re_g$, and $We_m$. Moreover, as the liquid film thickness increases, it converges to Wongwises and Kongkiatwanitch’s (2001) predictions. In the suction line of air-conditioning systems, if $\delta/D \geq 0.1$ then Wongwises and Kongkiatwanitch’s (2001) correlation can be used to estimate the interfacial friction factor, while if $\delta/D < 0.1$ then the new correlation, Equation 25, can be applied to approximate the interfacial friction factor in a horizontal suction line for both conventional refrigerants and CO₂. It should be noted that the model is unbounded as the liquid film goes to zero thickness. As the liquid film thickness decreases, the surface roughness of the pipe wall becomes important, and the model does not consider this effect. Thus, the model is valid only for liquid film thicknesses greater than zero and for smooth tubes. Moreover, for similar mass flow rates, a thinner liquid film has greater velocity gradient because it goes from zero velocity at the pipe wall to a maximum velocity at the gas-liquid boundary interface in a small thickness. The viscous shear stress
in a thinner liquid film is proportional to its velocity gradient so the resistance becomes higher in a thinner liquid film. As a result, thinner liquid films are more difficult to carry over with the refrigerant vapor, and higher friction factors are expected as emphasized in the trend of Figure 5.

The experimental versus the correlated friction factors according to Equations 24 and 25 are plotted in Figure 7. The final results show that the correlations predicted the friction factor within an average relative error of about ±26%.

Equation 25 represents the most general equation that can be adopted to predict oil retention in the horizontal suction line of air-conditioning systems. It has been developed and verified in the following ranges:

- \( 0.001 \leq \delta/D < 0.06 \)
- \( 17,000 \leq Re_g \leq 40,000 \)
- \( 13.8 \leq We_m \leq 221 \)
- \( 20 \leq \nu_l/\nu_g \leq 48 \)

These represent the widest possible ranges of liquid film thicknesses, refrigerant vapor Reynolds numbers, mixture Weber numbers, and mixture viscosity ratios for most common refrigerant/oil mixtures in air-conditioning systems.

Figure 8 compares experimental oil retention volume (Cremaschi 2004) versus predicted oil retention volume, which was obtained by using Equations 24 and 25 to close the problem specified previously by the system of transport equations. The simulation results were within ±31% average relative error. The friction factor correlation (Equation 25) was used only for horizontal suction lines while, for vertical suction lines, the friction factor correlation (Equation 24) provides more accurate results. However, the specific friction factor correlation for upward vertical suction lines has been obtained only for the R-22/MO mixture, and its accuracy has not been verified for other refrigerant/oil mixtures. In the future, more studies should focus on the correlations for vertical suction lines and on verifying them for different oil and refrigerant mixtures.

**Figure 7. Experimental versus correlated interfacial friction factor in the suction line.**


Oil Retention in the Evaporator

Figure 9 shows an example of the calculated air temperature and mixture bubble temperature profiles versus dimensionless length of the heat exchanger. The refrigerant/oil mixture was R-134a/PAG and the inlet saturated pressure of the evaporator was about 465 kPa. The inlet refrigerant quality was \( x_{\text{ref}} = 0.06 \) while the OMF was about 2.4 wt.%. Thus, the inlet mixture vapor quality was calculated using the following relation:

\[
x_{\text{mix}} = x_{\text{ref}} \cdot (1 - \text{OMF})
\]  

(26)

Figure 8. Experimental versus predicted oil volume retained per unit length of suction line.

Figure 9. Calculated air and bubble temperatures in the evaporator from the simulation results of R-134a/PAG at OMF = 2.4 wt.%.
In Figure 9, the vapor quality ranges from \(0.058 \leq x_{\text{mix}} \leq 0.96\), while the maximum possible quality is \(x_{\text{mix,max}} = 0.976 (= 1 \text{ – OMF})\) due to the presence of oil that does not evaporate. The actual effectiveness of the heat exchanger and the approach temperature were computed by the \(\varepsilon\)-NTU method existing in the simulation tool. For this specific example, the effectiveness of the heat exchanger was about 95\% and the approach temperature was 3 K. The figure shows that the rise in the bubble-point temperature limits the fraction of the refrigerant that can actually be evaporated because the temperature difference between the hot and cold sides decreases. The pressure drop across the heat exchanger was about 14 kPa and the predicted mixture outlet temperature was about 16°C (61°F). It should be noted that if the pressure is assumed constant, the bubble temperature increases with the vapor quality, as shown in Figure 9 by the bubble temperature line during evaporation without pressure drop. However, if the pressure drop is included in the model, the bubble temperature first decreases due to lower saturation pressure and then increases at the end of the evaporator. In this specific example, the number of tubes was 14 and the number of segments for each tube was 16. Thus, the total number of segments along the curve was 224. If the number of segments per tube was increased to 30, the bubble temperature profile changed by less than 0.1 K. For a number of segments per tube higher than 16, the sensitivity of the bubble-point temperature was less than or equal to 0.1 K, the variation of oil retention was less than 0.01 mL/m, and the predicted pressure drops across the evaporator did not change.

Figure 10 shows the distribution of oil retention and vapor quality with respect to the dimensionless length of the evaporator. The calculated values are for the specific example discussed previously with R-134a/PAG at OMF = 2.4 wt.%. The oil retention is given on the primary y-axis in dimensionless form, that is, as a ratio between the local oil retention volume over the maximum oil retention volume in the circuitry (\(\text{max } OR_{\text{segm}} = 0.07 \text{ mL})\). The mixture quality is given on the secondary y-axis, and the length of the circuitry is measured along the x-axis in

![Figure 10](image.png)

Figure 10. Calculated oil retention, vapor quality, and local oil mass fraction in the evaporator (R-134a/PAG at OMF = 2.4 wt.%).
The dimensionless oil retention was about 0.6 at the inlet mixture quality of about \( x_{\text{mix}} = 0.06 \), where almost all the refrigerant was in the liquid phase. Along the heat exchanger, the liquid refrigerant started to evaporate, leaving a higher percentage of pure oil in the liquid solution. In the first part of the evaporator, the oil retention decreased because the liquid refrigerant evaporated and the local oil mass fraction was low. Thus, the liquid film thickness decrease was the predominant effect. As the local oil mass fraction started to rise, the dimensionless oil retention increased rapidly from 0.4 to 0.8 and reached the maximum value at the evaporator outlet (the dimensionless oil retention at the evaporator outlet is equal to 1). Most of the oil was retained at the end part of the evaporator, where the liquid film viscosity was 2.8 cSt, the highest value in the evaporator. In the graph of Figure 10, it should be noted that the local air temperature affected the rate of increase of the mixture quality. The two kinks occurred exactly at the transition between one column and the next in the circuitry due to a decrease of the air temperature. Moreover, in counterflow configurations, the warmest air entering the heat exchanger caused the mixture quality rate of change to be highest in the last tube bank of the refrigerant circuitry.

The oil retention simulation results for the evaporator are shown in Figure 11 for all tests conducted in the experimental part of this research. The x-axis gives the predicted oil retention volume from the simulations, while the measured oil retention volumes are plotted on the y-axis. The continuous curve represents the predicted oil retention volume according to Premoli’s void fraction model, while the data points are the corresponding measured oil retention volumes for all refrigerant/oil mixtures tested. The approach according to Premoli’s void fraction model provides promising results, as shown in Figure 11. The average deviation was about 21% and the standard deviation was 15%. For the highest oil retention volumes, associated with the R-410A/MO mixture, if the average bulk liquid film viscosity and surface tension are employed in the model, then the simulations always underestimate the measured oil retention volume by

![Figure 11. Experimental versus predicted oil retention volume in the evaporator.](image-url)
more than 50%. When local properties according to Equations 19 and 20 are used in the correlations instead of bulk film properties, the accuracy improvement of the predicted oil retention volumes for the R-410A/MO mixture was more than 50%. Overall, 72% of the experimental oil retention volumes were predicted within ±29% relative error by the heat exchanger design software for all refrigerant/oil mixtures tested.

CONCLUSIONS

Numerical semi-empirical models for estimating oil retention in the suction line and evaporator were developed for R-22, R-410A, and R-134a air-conditioning systems. The modeling approach consists of an engineering approach that is both simple and sufficiently accurate to be useful in practical applications. The conclusions of this paper are as follows:

- New semi-empirical correlations for predicting the interfacial friction factors in horizontal and upward vertical refrigerant/oil annular flows in the suction line were developed for practical use. The new correlation was verified for different refrigerant/oil mixtures and for the widest possible range of transport properties in air-conditioning systems. The correlations suggested that the interfacial friction factor is inversely proportional to the dimensionless liquid film thickness, refrigerant vapor Reynolds number, and mixture Weber number.
- The new correlations for interfacial friction factor are in reasonable agreement with the CO2/PAG oil retention experimental results from Lee’s (2002) research, and they converge to the predictions of Wongwises and Kongkiatwanitch (2001) as the dimensionless liquid film thickness approaches 0.1. The relative average error between numerical and experimental friction factors was ±26%. Oil retention volume in the suction line was estimated by assuming a constant liquid film thickness along the entire pipeline. The predicted oil retention volume was within ±31% average relative error compared to experimental data.
- If the dimensionless liquid film thickness δ/D ≥ 0.1 then Wongwises and Kongkiatwanitch’s (2001) correlation can be used to estimate the interfacial friction factor, while if δ/D < 0.1 then the new correlations presented in this paper can be applied to approximate the interfacial friction factor for both conventional refrigerants and CO2.
- A comprehensive, semi-empirical model was developed for the evaporator. The refrigerant and oil heat transfer and pressure drop correlations available in the literature were implemented in a simulation tool to model the heat exchangers. These correlations were verified with experimental data. While the Lockhart-Martinelli pressure drop correlation poorly predicted the two-phase pressure drop in the evaporator, the cooling capacity of the system was computed within 12% relative error.
- Premoli’s void fraction model agreed closely with the experimental data, and oil retention in the entire evaporator was estimated within 21% average relative error. About 72% of the data points were within 29% relative error.
- In the evaporator, oil retention volume reaches a maximum value at the end part of the heat exchanger since the local liquid film viscosity is the highest. In refrigerant/oil mixtures, the liquid film viscosity depends on the mixture temperature and the local oil mass fraction in the liquid film.

ACKNOWLEDGMENTS

The authors would like to acknowledge the support of the sponsors of the Center for Environmental Energy Engineering (CEE) at the University of Maryland.
NOMENCLATURE

\( A = \text{area, m}^2 \)
\( C = \text{constant} \)
\( c_p = \text{specific heat at constant pressure, J/(kg} \cdot \text{K)} \)
\( D = \text{diameter, m} \)
\( \text{EES} = \text{Engineering Equation Solver} \)
\( F = \text{friction factor} \)
\( g = \text{gravity, m/s}^2 \)
\( G = \text{mass flux, kg/(m}^2 \cdot \text{s)} \)
\( H = \text{convective transfer coefficient, W/(m}^2 \cdot \text{K)} \)
\( K = \text{thermal conductivity, W/(m} \cdot \text{K)} \)
\( L = \text{length, m} \)
\( M = \text{mass, kg} \)
\( \text{MO} = \text{mineral oil} \)
\( N = \text{empirical exponent} \)
\( \text{OMF} = \text{oil mass fraction} \)
\( OR = \text{oil retention, ml} \)
\( \text{ORM} = \text{oil retention mass, g} \)
\( P = \text{pressure, Pa} \)
\( \text{PAG} = \text{polyalkylene glycol oil} \)
\( \text{POE} = \text{polyolester oil} \)
\( r = \text{radial coordinate} \)
\( R = \text{radius, m} \)
\( \text{Re} = \text{Reynolds number} \)
\( S = \text{solubility} \)
\( \text{UA} = \text{overall heat transfer coefficient, W/K} \)
\( \text{We} = \text{Weber number} \)
\( X = \text{x-coordinate in Baker’s map} \)
\( x = \text{flow quality} \)
\( \text{X}_l = \text{Lockhart-Martinelli parameter} \)
\( Y = \text{y-coordinate in Baker’s map} \)
\( Z = \text{axial coordinate} \)

Greek Symbols

\( \alpha = \text{void fraction} \)
\( \delta = \text{liquid film thickness, m} \)
\( \eta = \text{fin efficiency} \)
\( \lambda = \text{property correction factor} \)
\( \mu = \text{dynamic viscosity, Pa} \cdot \text{s} \)
\( \nu = \text{kinetic viscosity, cSt} \)
\( \rho = \text{density, kg/m}^3 \)
\( \sigma = \text{surface tension, N/m} \)
\( \tau = \text{shear stress, Pa} \)
\( \omega = \text{concentration} \)
\( \psi = \text{property correction factor} \)

Subscripts

\( a = \text{air} \)
\( c = \text{core} \)
\( g = \text{gas} \)
\( i = \text{interfacial} \)
\( l = \text{liquid} \)
\( o = \text{oil} \)
\( \text{ref} = \text{refrigerant} \)
\( \text{tot} = \text{total} \)
\( w = \text{water} \)

REFERENCES


**APPENDIX A**

The viscosity of the liquid film is the most important parameter affecting oil retention volume. More viscous liquid films are more resistant to the refrigerant driving force and are therefore more difficult to carry. Various combinations of the refrigerant and oil mixture provided a wide range of liquid film viscosities and densities. The degree of solubility and miscibility between the oil and refrigerant affected the thermodynamic properties of the liquid film. Density and viscosity of the liquid film have been computed according to temperature and pressure measurements and solubility effects (ASHRAE 2002). The thermodynamic properties of the oil and refrigerant mixtures were provided by the refrigeration companies that sponsored the research.
work at CEEE, University of Maryland. Since they were available only in graphic form, surface-fit interpolations were necessary. All the predicted values from the surface interpolations were within 90% confidence limit with respect to the measured data. For the mixture R-22 and MO, a mixing rule based on the component mole fraction and molecular mass was used because of the limited experimental data available. This rule was recommended by Yokozeki (1994).

For the R-22/MO mixture, the dynamic viscosity of the liquid film was between 10 and 22 mPa·s at a pressure of about 500 kPa and temperatures that ranged from 5°C to 30°C (41°F to 86°F). The average oil film solubility ranged from 6% to 25% in mass in the suction line, depending on pressure and temperature. The kinetic viscosity of the liquid film of MO with R-22 dissolved in it ranged from 11 to 25 cSt.

The effect of increased mutual miscibility between oil and refrigerant on the oil retention volume reduction is quite difficult to predict. For R-410a the condensation and evaporation pressures were about 2370 kPa and 1010 kPa, respectively. The degree of subcooling was 5 K for R-410A/MO and 6 K for R-410A/POE. The degree of superheat was 18 K for R-410A/MO and 22 K for R-410A/POE. The average solubility of R-410A in MO oil in the suction line was about 6 mass %, while the solubility of R-410A in POE oil was 11 mass %. The estimated average liquid film viscosity in the suction line was 28 cSt for R-410A/MO and the actual liquid film viscosity of R-410A/POE was 22 cSt.

More soluble oil refrigerant pairs, such as R-134a/POE and R-134a/PAG, have reduced liquid film viscosity compared to R-22/MO. POE and PAG synthetic oils had different degrees of solubility and miscibility with R-134a. The PAG oil was ISO VG 46 grade lubricant while the POE oil viscosity was slightly less (ISO VG 32). The average PAG oil solubility in the suction line was about 28 mass % and the POE oil solubility was only 21 mass %. As a result, the liquid film viscosities in the data were very similar for both mixtures and equal to approximately 14 cSt for R-134a/POE and 15 cSt for R-134a/PAG.

Finally, the surface tension of a generic pure mineral oil is equal to about 33 mN/m in air. The surface tension of POE oil is about 46 mN/m, while the surface tension of PAG oil is about 31.5 mN/m. These are common values for compressor lubricants used in different air-conditioning applications, but they can vary depending on the chemicals added to the oil. The dynamic surface tensions of the liquid films of oil with refrigerant dissolved in it were estimated according to the method recommended by Shen and Groll (2003).

APPENDIX B

An empirical vapor pressure equation for predicting the bubble-point temperature for a given saturation pressure and oil mass fraction was presented by Thome (1995) for different refrigerant and oil pairs. If $P_{sat}$ is the saturation pressure in MPa and $T_{bub}$ is the bubble-point temperature in K, the relation is given by

$$T_{bub} = \frac{A(\omega_{local})}{\ln P_{local} - B(\omega_{local})}, \quad \text{(B-1)}$$

where $\omega_{local}$ is the local oil mass fraction in the liquid, $P_{local}$ is the local pressure in each segment of the tube, and $A(\omega_{local})$ and $B(\omega_{local})$ are given by the following expressions:

$$A(\omega_{local}) = a_0 + a_1 \omega_{local} + a_2 \omega_{local}^3 + a_3 \omega_{local}^5 + a_4 \omega_{local}^7$$

$$B(\omega_{local}) = b_0 + b_1 \omega_{local} + b_2 \omega_{local}^3 + b_3 \omega_{local}^5 + b_4 \omega_{local}^7$$
Because the vapor pressure of the lubricant is negligible, there is not a large difference between the lubricant partial pressures of different refrigerant-lubricant pairs. Consequently, Thome recommended correlating \( a_0 \) and \( b_0 \) with the specific pure refrigerant while keeping the other constants unchanged. The values of the other empirical constants are:

\[
\begin{align*}
    a_1 &= 182.52 & a_2 &= -724.21 & a_3 &= 3868.0 & a_4 &= -5268.9 \\
    b_1 &= -0.72212 & b_2 &= 2.3914 & b_3 &= -13.779 & b_4 &= 17.066
\end{align*}
\]

The constants \( a_0 \) and \( b_0 \) depend on the specific refrigerant and saturation pressure. They were computed as follows:

- The inlet pressure of each segment of the evaporator was set as \( P_{local} \).
- Two saturation temperatures just above and below \( P_{local} \) were determined using an accurate equation-of-state relation for the pure refrigerant vapor pressure curve.
- The two sets of values for \( T_{bub} \) and \( P_{local} \) were set in Equation B-1 with \( \omega_{local} \) set to zero.
- The system of two equations in two unknowns \( (a_0, b_0) \) was solved. All other values of \( a_1 \) and \( a_4 \) and \( b_1 \) and \( b_4 \) remain the same as in the original correlation since they only refer to the effect of oil on \( T_{bub} \).

The local change in enthalpy \( dh \) of a mixture during evaporation consists of three contributions as stated by Thome (1995):

1. latent heat to the fraction of the liquid vaporized \( (dx_{mix}) \);
2. sensible heat to the fraction of fluid in the liquid phase \( (1 - x_{mix}) \) heated to a higher bubble-point temperature;
3. sensible heat to the fraction of fluid in the vapor phase \( (x_{mix}) \) heated to a higher bubble-point temperature.

Thus, \( dh \) can be written as follows:

\[
    dh = h_L V \cdot dx_{mix} + (1 - x_{mix}) \cdot c_{p,mix} \cdot dT_{bub} + x_{mix} \cdot c_{p,ref,vapor} \cdot dT_{bub}
\]

The values of \( c_{p,mix} \) and \( c_{p,ref,vapor} \) are functions of the local oil mass fraction and bubble-point temperature while \( h_L V \) is a function of local pressure. The equation neglects the mixing heats, which are normally small and typically not available for refrigerant/oil mixtures. The above equation is called the heat release enthalpy curve, and it reduces to the latent heat of vaporization for a pure refrigerant.